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Screw pump

This invention relates to the field of vacuum pumps. In particular thermal control of vacuum pumps with a screw type configuration.

Screw pumps usually comprise two spaced parallel shafts each carrying externally threaded rotors, the shafts being mounted in a pump body such that the threads of the rotors intermesh. Close tolerances between the rotor threads at the points of intermeshing and with the internal surface of the pump body, which acts as a stator, causes volumes of gas being pumped between an inlet and an outlet to be trapped between the threads of the rotors and the internal surface and thereby urged through the pump as the rotors rotate.

Prior art screw pumps use a water cooling jacket around sections of the machine in order to remove the heat of compression. However, the inlet of the machine does not have any cooling system since, at low pressures, there is little heat of compression to be removed from the inlet. As the pressure increases any additional heat is dispersed from the inlet by the increased gas flow through it. Where the pump is located in a cold environment, surface temperatures within the inlet of the pump may reduce significantly and form cold spots such that gaseous waste products from the evacuation chamber condense into liquid pools in these cooler regions. These pools can be formed from highly corrosive acid or base fluids and can lead to damage of the pump components, which, in turn, can reduce the life of the device.

Double ended screw pumps are known where a single inlet serves two outlets, the rotors being mounted in a co-linear fashion. In such a pump a disparity in temperature between the inlet and the outlet sections of the pump is more pronounced and concentricity of the bores within the housing components becomes important. If the housing components move out of alignment the rotor is more likely to clash with the stator as the already small tolerances reduce even further or are eliminated.

Screw pumps are increasingly being utilised in a broad range of applications. For example within a pharmaceutical process area the same pump may be required to perform numerous different applications. Whilst the configuration of a pump may be tailored to a particular application, once the application is altered, ideal conditions

will no longer be present and the pump will not be performing at peak/optimum efficiency.

It is an aim of the present invention to overcome some of the aforementioned problems associated with screw pump technology.

According to one aspect of the present invention there is provided a pump comprising:

a stator;

at least one rotor mounted within a housing, the housing comprising a first fluid channel extending about the rotor, the rotor comprising at least one second fluid channel;

a first sensor configured to output a signal indicative of the temperature of the stator;

a second sensor configured to output a signal indicative of the temperature of the rotor; and

thermal control means for controlling the temperature of fluid, when present, in said channels depending on the magnitude of signals output from the sensors.

The first temperature sensor may be located at the stator, whereas the second temperature sensor may be located either in the exhaust plenum or within the housing, in fluid communication with process gas in an exhaust portion of the rotor, alternatively it may be situated in the gear box of the pump

The thermal control means may comprise first and second control means for controlling the temperature of any fluid in the first and second channels respectively. Either thermal control means at least one of each of a variable speed flow pump, a thermostatic control valve and a heat exchanger. They may be arranged to control the temperature of the fluid in the respective channels dependant on the magnitude of one or more of the sensors' outputs. The thermal control means may include/be controlled by a microprocessor.

One of the thermostatic control valves may comprise a mechanical differential temperature valve. This valve may comprise a third fluid channel in thermal communication with the second fluid channel. A flow restrictor may be provided within this third fluid channel to control the rate of fluid therethrough. The position of this flow restrictor may be governed by signals received from the first and second

sensors via signal receptors which may also form part of the valve. Each signal receptor may comprise a sealed component, the volume of which may expand in use. The degree of expansion being dependent on the magnitude of the signal received and determining the relative position of the restrictor within the third fluid channel. The sealed component of the signal receptor may comprise an expandable bellows. The flow restrictor may comprise a spindle and a seat. The spindle acting co-operatively with the seat to open and close an aperture to control the flow of fluid therethrough.

The pump may be of any known form, for example but not strictly limited to; a screw pump, a claw pump or a Roots pump.

According to a further aspect of the present invention there is provided a double-ended pump comprising at least one rotor, comprising:

one inlet portion and two exhaust portions;

a stator; and

a housing, the housing comprising an inner skin and an outer skin, a first cavity being formed by the inner skin, the rotor being mounted therein and a second cavity being formed between the inner and outer skins of the housing through which a fluid is circulated, in use, wherein the second cavity extends the length of and encircles the rotor.

According to a further aspect of the present invention there is provided a valve comprising:

a fluid channel;

a flow restrictor moveable within the fluid channel to control the rate of flow of a fluid therethrough; and

two signal receptors for receiving respective signals and controlling the position of the flow restrictor within the channel depending on the magnitude of the received signals.

According to a further aspect of the present invention there is provided a method for releasing the rotors of a pump that have seized due to the presence of deposits of a substance which has solidified on the internal working surfaces of the pump on cooling, comprising the steps of:

introducing a thermal fluid into a cavity provided within the housing of the pump, the cavity encircling the rotor components;

heating the thermal fluid in the cavity to a predetermined temperature, this temperature being sufficiently high to cause the deposits to be softened; and applying torque to the rotors of the pump to overcome any remaining impeding force caused by the deposits located on the internal working surfaces of the pump.

According to a further aspect of the present invention there is provided a method for controlling a clearance between a rotor and stator within a pump, of the present invention, the method comprising the steps of:

(a) recording the temperature of each of the stator and the rotor from the sensors;

(b) calculating the temperature difference between the stator and the rotor;

(c) comparing the temperature difference with a predetermined value;

(d) determining suitable values of flow rate and temperature for the fluid in the first and second fluid channels to achieve the predetermined temperature difference; and

(e) controlling the thermal control means to realise the values from step (d).

The method steps may be repeated automatically at predetermined time intervals in order to manage perturbations in the configuration of the pump over time. The predetermined temperature difference may be modified at predetermined time intervals to cause the clearance between components to be altered such that cumulative deposits can be dislodged from the surfaces of the components of the pump.

The thermal controller may comprise a microprocessor which may be embodied in a computer, which in turn is optionally programmed by computer software which, when installed on the computer, causes it to perform the method steps (a) to (e) mentioned above.

The present invention enables a pump to be subject to an improved level of thermal control. This allows benefits to be achieved during operation of the apparatus in terms of providing optimised running clearances enhancing the tolerance of the pump to excessive exhaust back pressures, reducing the occurrence of cold spots in the inlet of the pump, reducing thermal lag in the apparatus and enhancing the likelihood of restart in circumstances where deposits are formed due to cooling.

An example of the present invention will now be described with reference to the accompanying drawings in which:

Figure 1 illustrates a schematic plan cross section of a screw pump of the present invention;

Figure 2 illustrates a plan cross section of a double-ended screw pump of the present invention;

Figure 3 is a schematic of a temperature control circuit of the present invention;

Figure 4 illustrates further detail of the interface between the rotor and the stator of the pump in Figure 2;

Figure 5 illustrates a more sophisticated version of the present invention;

Figure 6 shows a further example of the present invention for use in more severe environments;

Figure 7 shows detail of a differential temperature valve for use in the pump of Figure 6: and

Figure 8 illustrates a Roots blower implementing thermal control of the present invention.

Screw pumps are illustrated in Figures 1 and 2. Two rotors 1 are provided within an outer housing 2. The two contra-rotating, intermeshing rotors 1 are positioned such that their central axes lie parallel to one another. The rotors 1 are mounted in the housing 2 via bearings 3. The single ended pump of Figure 1 comprises an inlet stator 4 and an exhaust stator 5, whereas the double ended pump of the example in Figure 2 comprises an inlet stator 4 positioned between two exhaust stators 5.

The housing 2 is provided as a double skinned construction. The internal skin acts as the stator of the pump. A cavity 6 is provided between the skins of the housing 2 such that a cooling fluid, such as water, can be circulated around the stator in order to conduct heat away from the working section of the pump. This cavity 6 encircles the full length of the stator i.e. over the inlet stator 4 as well as the exhaust stators 5. Cooling fluid is circulated through this cavity to draw heat away from the hot surface. By providing the water jacket over the length of the stator, heat generated towards the exhaust end of the rotor can be redistributed to the earlier stages when necessary. This will enable the temperature gradient to be reduced and allow a more uniform temperature to be maintained over the surface of the pump. Consequently, the 'cold spots' found in the prior art can be avoided and condensation

of potentially corrosive materials in the rotor inlet are substantially reduced. Furthermore, thermal lag is introduced into the system due to the presence of a complete water jacket which effectively prevents rapid temperature changes in the stator and rotor surfaces e.g. through wind chill effects. The maintenance of a uniform temperature causes all of the stator components to expand at the same rate from a central datum (the shaft) thus concentricity can be maintained and, consequently, the rotor retains its relative position within the stator and clashing of components can be avoided.

Conventional pumps with thermal jackets generally use convection to circulate the thermal liquid through the stator. This can lead to uneven distribution of temperature over the pump, notably, with cooler areas lower down the pump and warmer sections in the upper regions. Such localised cold spots can cause the process gases to condense out, becoming increasingly corrosive. By implementing a circulation pump within the thermal fluid consistent thermal control can be achieved such that local variations in temperature can be minimised.

In some cases, the waste products passing through the pump comprise a waxy or highly viscous substance and deposits are formed on the surfaces of the pump during operation. On shut down of the pump, these deposits cool and may solidify. Where such deposits are located in clearance regions between components, they can cause the pump to seize up. The motor may then provide insufficient torque to overcome this additional friction and cause the rotor to rotate. Additional torque can be applied using a leverage bar inserted into a socket on the shaft, which can then be rotated manually. However such a technique exerts a significant load on the rotor and may cause damage. However, it may not be possible to exert sufficient load to release the mechanism and force the shaft to rotate, under these circumstances it may be necessary to decommission the apparatus and take it out of service either for replacement or repair. An alternative use of the water jacket of the present invention can be implemented in these circumstances where the pump has become seized due to cooling of the rotor. The fluid in the cavity 6 of the housing 2 may be heated to raise the temperature of the stators 4,5 and the rotors 1. This can enhance the pliability of the residue and may assist in releasing the mechanism.

Figure 3 shows how fluid circuits 11,12, 12a and 15 may be used to control thermal conditions within the pump. The cooling liquid, typically a mixture of water and anti-freeze, is provided in a first closed circuit 11 with a circulation pump 17. A second fluid circuit 12, comprises a pressurised water circuit and a thermostatic

control valve 13. Typically, mains water is provided to this circuit at inlet 25 and is removed at outlet 26. A heat exchange component 14 is provided between these two fluid circuits 11, 12. The valve 13 receives an input signal from a thermal sensor 21 located at the stator and uses this to maintain a suitable flow rate in the second circuit 12 to govern the temperature gradient over the heat exchange component 14. This temperature gradient, in turn, maintains the temperature of the first circuit 11.

The rotor 1 comprises a threaded section 9 and a separate shaft component 8 as illustrated in Figure 2. The threaded section 9 is provided with an internal cooling cavity 7, into which is inserted the body of a separate shaft 8. The body of shaft 8 is fractionally smaller in diameter than the diameter of the cooling cavity 7 within the body of the rotor. Thus a cooling channel is provided, through which a coolant material, typically oil, may be passed. This channel is kept small to encourage the flow speed of the coolant to be as high as possible, in order to enhance the cooling function by maintaining the temperature difference between the rotor and the coolant and transporting heat back to a coolant reservoir. The cooling system inlet and outlet are provided through the shaft component 8.

Returning to Figure 3, it can be seen that the oil is retained in another closed circuit 15. This circuit 15 comprises a circulation pump 19 a filter 20 and a heat exchange component 14a. This heat exchange component is in contact with a second cooling circuit 12a and comprises a further thermostatic control valve 16. The thermostatic control valve 16 receives an input signal from a second thermal sensor 22 which indicates the temperature of the rotor either via the oil from circuit 15 or through the process gases within the latter stages of the rotor. Temperature may be monitored within the exhaust plenum (i.e. the cavity between the end of the rotor and the stator wall) but here, the temperature will rapidly fall below that of the rotor.

By introducing two thermostatic control valves as described, it is possible to control the temperature of the pump rotor relative to the stator temperature. The rotor to stator clearance d (as illustrated in Figure 4) in a screw pump is a function of this difference in temperature between the rotor and the stator. By controlling the temperatures of these components it is possible to control the magnitude of this clearance d. Furthermore, it is this clearance d that determines the level of leakage of process gas between the rotor and the stator, such that the volume of leakage is proportional to d^3 for intermediate, transitional flow as shown in "Modern vacuum practice" by Nigel Harris, McGraw Hill (p231). Since leakage affects the performance of the pump, it follows that the performance of the pump can be optimised by

controlling the temperatures of these components. Furthermore, it is beneficial to be able to maintain clearances d such that any particulate content of the process gas does not form a blockage within these clearances and thus inhibit the free running of the rotor 1. Such obstruction can severely affect the performance of the pump through restriction of through flow of the process gas but also through additional torque that must be applied by the motor in order to maintain the appropriate speed of rotation of the rotor.

By providing a temperature control circuit within the rotor it is possible to thermostatically control rotor temperature relative to the stator temperature to optimise rotor/stator clearance d. In its simplest implementation the present invention can be used simply to avoid cold spots and thus eliminate corrosion due to condensation build up as discussed above. In a more sophisticated implementation, input signals can be taken from sensors mounted on each of the stator 4, 5 and the rotor 1 and these signals can be analysed/processed by a closed loop control system to maintain a set temperature, for example to be less than 135 °C. This allows a pump using the present invention to safely process materials with a known auto-ignition temperature.

However, as discussed above, the present invention can be used at an even higher level of sophistication to select particular temperatures that will result in a particular clearance d being achieved and maintained. Figure 5 illustrates how a processor or central processing unit (CPU) 27 may be incorporated into the system to receive signals indicative of the rotor and stator temperatures from sensors 22a and 21a respectively. These signals provide input to allow the processor 27 to determine the required temperatures of each of the components. The processor 27 then controls electronically activated valves 13a and 16a to provide a suitable level of coolant fluid to the heat exchange components 14, 14a to achieve the required thermal balance and subsequent clearance d.

Under normal steady state operation, a dry pump will attain a particular pumping speed determined by the clearance between the rotor and the stator. If the inlet pressure to the pump is increased more gas will enter the pump. This additional gas will cause the rotors to cool down with respect to the stator and hence the clearance d between these two components will increase. It follows that, at higher pressures, a significant amount of leakage around the rotor will occur. This is particularly problematic when pumping gas species such as helium, which typically result in low pump speeds and gas throughput being achieved when approaching

atmospheric pressures. With the control feature of the present invention it is possible to artificially reduce the clearance d between the rotor and the stator. Consequently leakage around the rotor may be reduced and the efficiency of the pump can be improved significantly. In the above example, when pumping helium, it is desirable to maintain a small gap to prevent leakage. However, the same pump could be used, in an alternative application, to pump Argon where a larger gap would be required. By providing a pump that can essentially be optimised during operation to function efficiently under varying conditions, a multi-purpose pump is achieved. This functionality can be used to good effect in fields, such as the pharmaceutical or chemical process industries, where a single pump needs to be used for different applications using the same tooling.

The temperature control can be dynamic within a particular process. On start up there will typically be a greater temperature difference since the temperature of the rotor increases at a faster rate than the stator due to the significant difference in thermal mass of the relative components. However, once the pump has reached a steady state this temperature difference will be reduced. By performing the temperature control dynamically, this early difference can be minimised such that the clearance d can be maintained at an approximately steady value. This, in turn, will lead to a more consistent level of pump efficiency.

The dynamic control of the clearances may be implemented in a cyclic manner when the pump is operating under normal conditions. At predetermined intervals the thermal conditions can be modified to reduce the clearances between the rotor and the stator for a short period of time. This will have the effect of removing/dislodging process deposits that have become adhered to these components. If this is repeated at intervals the cumulative build up of solid matter on the internal surfaces of the pump can be substantially reduced thus preventing seizure of the pump.

Seizure of the pump may be further be avoided by provision of an additional sensor for monitoring either the pressure within the pump or the power consumption of the pump. If either of these values increase significantly, this may be an indication that the clearances are becoming obstructed and that seizure is imminent. By monitoring these values it is possible to initiate a condition of maximum cooling of the rotor component to maximise the clearance between the rotor 1 and the stator 4, 5 and thus prevent seizure of the pump.

Alternatively the thermal control means may be provided by a purely mechanical means as illustrated in Figures 6 and 7 where a particular temperature difference can be automatically maintained between the stator and the rotor. In this way a simpler but more robust device can be implemented in pumps that are exposed to particularly harsh conditions. The mechanical thermal control device 24 is directly connected to a sensor 22 located as described above to indicate the rotor temperature through the process gas within the swept volume or oil temperature within the gearbox and also to a sensor 23 located within the stator of the pump. This latter sensor 23 may be located in a similar position to sensor 21 which provides input to the thermal control valve 13 in Figure 3. Each end of the differential temperature valve experiences a different temperature from each sensor causing a sealed sensor/bellows arrangement to be heated thus causing an expansion of the bellows. These two expanding bellows arrangements act in combination to position an internal valve. This valve position governs the amount of cooling fluid that may pass through the thermal circuit and thus alters the heat removal of the heat exchange unit 14a. This, in turn, controls the clearances within the pump by modifying the temperatures of the pump components. This simpler example maintains a temperature difference between the rotor 1 and the stator 4, 5 rather than actively controlling each component individually. However, by maintaining these relative temperatures a consistent clearance can be maintained. The valve 24 can be physically altered, for example, by restricting the expansion of one of the bellows components to adjust the magnitude of temperature difference between the rotor 1 and the stator 4, 5, thus allowing different processes to be accommodated.

The present invention is not restricted for use in screw pumps and may readily be applied to other types of pump such as claw pumps or Roots pumps. Indeed in some Roots blowers, significantly higher exhaust pressures (in some cases up to 2 to 3 bar) can be experienced. These raised pressures lead to a notable increase in component temperatures within the pump which can, in turn, lead to problems in maintaining appropriate clearances. By implementing dynamic thermal control according to the present invention these clearances can be maintained at consistent levels thus improving the tolerance of the pump to different operating conditions.

A rotor 35 from a Roots blower is illustrated in Figure 8, in order to introduce the thermal control of the present invention it is necessary to introduce a cooling channel 34 into the rotor 35 in a similar manner to that found in the screw rotor 1 of

Figure 2. Once again the channel is kept small to encourage the flow speed of the coolant to be as high as possible, in order to enhance the cooling function by maintaining the temperature difference between the rotor 35 and the coolant and transporting heat back to a coolant reservoir, typically the gear box (not shown). The cooling channel inlet 32 and outlet 33 are provided through the rotor shaft component 31. The cooling channel passes into each of the lobes 30 present on a Roots rotor 35, there may be two lobes as illustrated or there could readily be three, four or even more lobes on the rotor.

It is to be understood that the foregoing represents just a few embodiments of the invention, others of which will no doubt occur to the skilled addressee without departing from the true scope of the invention as defined by the claims appended hereto.